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Determination of the thermal stress state for the composite brake pad of a wagon at operational loads

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Abstract. The article provides the results of determining the thermal stress state of the composite brake pad of a wagon. The research is made by means of computer simulation with the finite element method using the SolidWorks Simulation options. The design diagram of the pad includes not only the horizontal loads due to its pressing to the rolling surface of the wheel and the friction force, but also the temperature impact taken equal to 400°C. Since the composite material of the pad is fragile, the Mohr–Coulomb criterion is used as a calculation criterion. It has been found that the maximum stresses occur in the upper part of the pad in the contact area between the back plate and the side plate and do not exceed the permissible values. The article presents the results of the calculation of the thermal stress state of the pad with dual wedge-shaped wear. The study includes the actual parameters of the pad wear determined during operational research. The results of the calculation show that the maximum stresses occur in the back plate of the pad and exceed the permissible values by 19.8%. This is explained by the fact that the useful area of the pad decreases, and therefore its loading increases. The research conducted proves the negative impact of dual wedge-shaped wear on the braking efficiency and the strength of the brake pad. This requires development of measures for eliminating this wear.

1. Introduction

The rapid development of the rail industry is accompanied by an increase of the train speed and the axle loading, the need to improve the structures and materials for the rolling stock, an increase in the tare load ratio of rail cars, etc. [1]. In this regard, friction materials used in the railway industry, in particular, for braking systems, should provide a fixed friction coefficient and low wear at various operational (speed, temperature, pressure) and environmental (noise, extreme weather) conditions.

The violation of the operational requirements causes an increased wear of brake pads (figure 1), and therefore, in operation brake shoes can be used for braking. This certainly leads to damage to the rolling surfaces of the wheels and directly affects the train traffic safety.

Therefore, in order to prevent abnormal wears of brake pads, including dual wedge-shaped wear, it is advisable to develop requirements and devices that will ensure the uniform wear of brake pads so that they can serve during the whole repair-to-repair period at depots. This makes it possible to dispose of such brake pads with a minor portion of the working composite mass remaining and save hundreds of thousands of hryvnias for Ukrainian Railways or industrial enterprises with their own freight rolling stock, so that they will purchase brake pads and significantly reduce operational costs.

(*a*) (*b*)

Figure 1. Consequences of abnormal wear of composite brake pads: (*a*) damaged brake shoe with wire instead of the key, used for braking without a pad; (*b*) part of the damaged pad remaining in the brake shoe.

The issue of traffic safety of freight trains is quite relevant and depends on many factors including the technical condition and the loading of structural components of their brakes. In study [2], for example, the authors analyse stresses and temperatures occurring in the brake pad by applying SolidWorks. They propose an alternative solution to use a composite based on modified alkyl benzene resin to increase the friction coefficient.

Koptovets et al [3] provides the results of testing on freight rolling stock of industrial transport in terms of the efficient braking, as well as structural and dynamic analysis of the brake mechanism. It includes the determination of the type and parameters of the empirical dependence between the friction coefficient of the brake pad on the rolling surface of the wheel and the braking speed, as well as the determination of the kinetic characteristic of the brake for freight rolling stock of industrial transport. However, the authors did not take into account dual wedge-shaped wear of the pads of wagons, which very significantly affects the braking efficiency of industrial transport.

Kiss et al [4] presents a new friction material for brake pads, which affects the service life of the wheels of rolling stock. Particular attention is paid to the problems related to the use of modern brake materials and their influence on thermal and mechanical properties during the transmission of loading to railway wheels when braking.

Muradian et al [5,6] presents the results of operational research into the assessment of factors causing defects of thermal origin on the rolling surface of the wheel pair when interacting with composite brake pads. To prevent such defects, the authors propose using composite pads with metal inserts that will reduce their wear. During the inspection of the brake equipment of the freight train, various malfunctions of the mechanical and pneumatic parts of the brakes were detected; the inspection of the brake pads demonstrated that their wear was wedge-shaped due to touching the upper end against the rolling surface. However, the works do not describe the impact of dual wedge-shaped wear on the strength of brake pads and braking efficiency.

Mazur et al [7] analyse the operational quality indicators of cast-iron and composite brake pads used on various types of rolling stock. The study describes some negative factors of composite pads and explains how they affect the environment and cause damage to rolling surfaces of wheels of rolling stock.

A lot of studies on the use of composite brake pads for rolling stock are dedicated primarily to the issues of traffic safety and environment protection. Therefore, the reduction in operational costs for the railway industry often means that brake pads are considered as a product that is often purchased at the lowest price if it performs satisfactorily. However, this may not imply the lowest operating costs, and the choice of friction material may have a direct impact on the service life of the wheel, the replacement of which is usually much more expensive than that of other car assemblies [8].

Mazur and Sirenko [9] compare the quality indicators and performance characteristics of cast-iron moulded and composite brake pads on the basis of publications overview. They also describe some disadvantages in using composite pads, such as, low thermal conductivity, which causes a thermal impact on the rolling surface of the wheel of rolling stock. This leads to an increase in the maintenance costs for wheel pairs. Another significant disadvantage is the fact that the manufacturing specifications, standards and technical documents do not include the list of components of the rubber mixture and their chemical composition; it contradicts the current legislation of Ukraine and makes it impossible to control these substances. However, the article does not mention the expenses caused by abnormal wear of composite brake pads, which can occur during movement of freight rolling stock when brakes are not applied.

Sharma et al [10] describe various friction braking devices used to reduce the resistance of movement. It is noted that friction brake mechanisms, in which composite brake pads are used, adversely affect the rolling surface of the wheel due to very high temperatures in the pad-wheel friction area, so preference is given to disc brakes.

Some scientists focus on studying disc brakes, calculating the strength of their elements, monitoring their operation, as well as calculating temperature modes for some parts of brake systems used for rolling stock [11, 12]. When friction brakes are applied, thermal energy is generated in the contact area of tribotechnical bodies; this energy is dissipated by forced convection, conduction and radiation from the exposed brake surfaces rotating during the train movement. Day et al [13, 14] state that overheated tribotechnical pairs can cause failures in the brake system, thus it can lead to violation of traffic safety. In this regard, significant theoretical research is being done to reduce the temperature during braking according to movement speeds and brake disc designs.

Other studies [15,16], aimed at introducing modern materials into the design of tribotechnical units, substantiate the effectiveness of their application in modern rolling stock; this makes it possible to increase the speed of movement, the axial load, the efficiency of the brake system, etc. But, at the same time, there are a number of problems associated with abnormal wear of brake pads in wagons that need to be solved. And the problems associated with wear of brake pads and wheels of freight rolling stock actually exist [17]. In this regard, the work related to improvements of the lever transmission elements of wagons to protect the movement of freight trains by increasing the efficiency of their brakes is being carried out.

The analysis of literature allows us to conclude that the issues of dual wedge-shaped wear of composite brake pads used in the brake systems of bogies in Ukraine at present are relevant and require research and development.

The objective of the research was to determine the thermal stress state of the composite brake pad used for a wagon at operational loads. To achieve this objective the following tasks were set:

- to investigate the thermal stress state of the composite brake pad with rated parameters;

- to investigate the thermal stress state of the composite brake pad with dual wedge-shaped wear.

2. The research into the thermal stress state of the composite brake pad with rated parameters

The thermal stress state of the brake pad at operation loading modes was determined using the strength calculation. It was carried out with the finite element methods in SolidWorks Simulation [18].

A composite brake pad 2TP-11 was chosen as the prototype. The main characteristics of the pad are given in table 1.

Table 1. The main characteristics of the brake pad.

Parameter	Measurement unit	Value
Overall dimensions		
– width	mm	80^{+2}_{-1}
– thickness	mm	$65^{+5}_{-1.5}$
Mass		
– for asbestos pad	kg	3.15 ± 0.2
– for asbestos-free pad	kg	3.2 ± 0.25

The spatial model of the pad was built in accordance with its album of drawings in SolidWorks (figure 2). The finite-element model of the pad was built with spatial isoparametic tetrahedrons with four Jacobian points (figure 3). The optimal number of elements of the model was determined by the graphoanalytic method. The mesh was based on the curvature. The number of elements in the mesh was 2829, and nodes – 12219. The maximum element size of the mesh was 15 mm and the minimal element size was 6.2 mm. The number of elements in the circle was 9. The element size gain ratio was 1.6.

The model was fixed by the back plate in the area adjacent to the shoe. The material of the pad was composite with linear elastic orthotropic properties. At the same time, the compressive

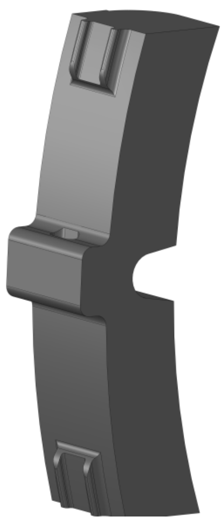

Figure 2. Spatial model of the brake pad.

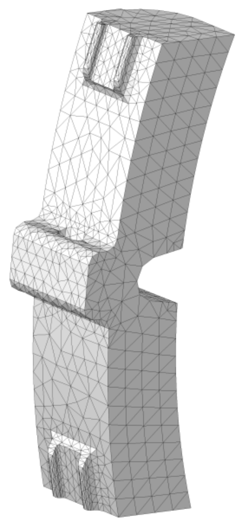

Figure 3. Finite-element model of the brake pad.

strength of the material was taken equal to 15 MPa, and the tensile strength was taken close to zero.

The design diagram of the pad is shown in figure 4. It included the horizontal load P_H to the working part of the pad, the value of which was taken according to the operational mode of the air distributor: 41.69 – freight mode; 34.34 – medium mode; 17.5 kN – empty mode [19]. Also, the model included the friction force P_{FR} determined by the formula:

$$P_{FR} = P_H \cdot \mu, \tag{1}$$

where μ – the friction coefficient ($\mu = 0.34...0.65$).

The authors took into account the average value of the friction coefficient $\mu = 0.5$.

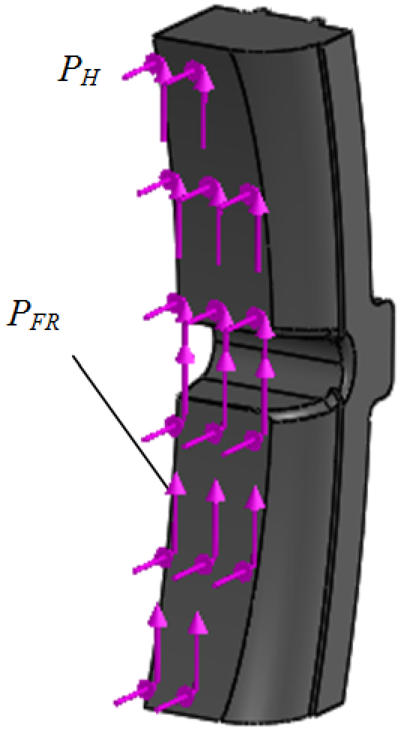

Figure 4. Design diagram of the pad.

It's worth noting that the pad suffered the temperature load during braking. The temperature on the rolling surface of the wheel in operation could be determined using the analytical expression [20]:

$$\Delta\tau_n = \frac{q_T}{\alpha_0} \cdot \left[1 - e^{-\frac{2\cdot\alpha_0}{\sqrt{\pi\cdot\lambda\cdot\gamma\cdot c}}\cdot\sqrt{t}\cdot\left(1-\frac{2}{3}\cdot\frac{t}{t_B}\right)}\right], \tag{2}$$

where q_T – the density of heat flux, kcal/(m^2 · °C); α_0 – the coefficient of heat transfer to the environment; λ – the thermal conductivity coefficient, kcal/(m^2 · °C); γ – the specific weight, kN/m^3); c – the specific heat capacity, kcal/(kgf · °C); t_B – the braking time until a complete stop, s.

The highest temperature during braking on the wheel surface was reached in the middle of this process $t = 0.5 \cdot t_B$:

$$\Delta\tau_{n\,\max} = \frac{q_T}{\alpha_0} \cdot \left[1 - e^{-0.9433\cdot\frac{\alpha_0}{\sqrt{\pi\cdot\lambda\cdot\gamma\cdot c}}\cdot\sqrt{t_B}}\right]. \tag{3}$$

The temperature on the surface of the wheel when the train stopped was $t = t_B$:

$$\Delta\tau_{nK} = \frac{q_T}{\alpha_0} \cdot \left[1 - e^{-0.667 \cdot \frac{\alpha_0}{\sqrt{\pi \cdot \lambda \cdot \gamma \cdot c}} \cdot \sqrt{t_B}}\right]. \tag{4}$$

The temperature as it was set during braking (at a constant speed):

$$\Delta\tau_{n\infty} = \frac{q_T}{\alpha_0} \cdot \left[1 - e^{-2 \cdot \frac{\alpha_0}{\sqrt{\pi \cdot \lambda \cdot \gamma \cdot c}} \cdot \sqrt{t_B}}\right]. \tag{5}$$

The following input parameters were used in the calculation: $q_T = 25.8$ kcal/(m^2 · °C); $\alpha_0 = 0.03$; $\lambda = 2 \cdot 10^{-4}$ kcal/(m^2 · °C); $\gamma = 2.2 \cdot 10^3$ kN/m^3; $c = 0.28$ kcal/(kgf · °C); $t_B = 120$ s.

By analysing dependencies (2) – (5), can conclude that the temperature on the pad surface was constantly changing by braking time. Therefore, when calculating the strength of the pad, the maximum allowable value of the temperature load P_T was taken into account. This load was applied to the working surface of the pad (figure 5) and was taken equal to 400 °C. The effect of this temperature on the pad material was estimated by introducing a coefficient of thermal expansion of $4.1 \cdot 10^{-6}$ K^{-1} for the material.

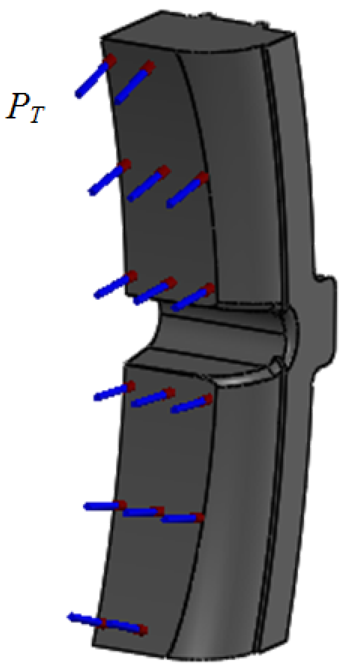

Figure 5. Diagram of the temperature loading applied to the pad.

Since the material mentioned had a small tensile strength and no yield strength, the strength calculation was made according to the Mohr–Coulomb criterion, i.e., the theory of internal friction. It is known that this criterion predicts failures if the simultaneous action of the maximum principle tensile stress and the minimum principle compression stress exceeds the appropriate stress limits [19].

In the case of a uniaxial stress state, the law of strength had the form [21]:

$$\tau \leq (\sigma - U) \cdot tg\varphi + c, \tag{6}$$

where τ and σ – the tangential and normal stresses acting at some point of the base; U – the pressure in the pore liquid; φ – the internal friction angle, c – the specific adhesion.

For the case of the spatial state, the formula took the form:

$$\frac{\sigma_1 - \sigma_3}{\sigma_1 + \sigma_3 + 2 \cdot c \cdot ctg\varphi} \leq \sin\varphi; \tag{7}$$

$$\sigma_1 > \sigma_2 > \sigma_3, \tag{8}$$

where σ_1, σ_2 and σ_3 – the principal stresses.

In accordance with this criterion, failures were predicted in the following cases:

- the principal tensile stresses were greater than zero $\sigma_1 > 0$ and $\sigma_3 > 0$. In this case, the failure criterion was taken into account if the principal stress exceeded the boundary tensile stress, i.e. $\sigma_1 > \sigma$;
- the principal compression stresses were less than zero $\sigma_1 < 0$ and $\sigma_3 < 0$. In this case, the failure criterion was taken into account if the admissible principal stress exceeded the boundary compression stress, i.e. $\sigma_1 > \sigma$;
- the principal tensile stress was $\sigma_1 > 0$, and principal compression stress was $\sigma_3 < 0$. The failure criterion in this case was:

$$\frac{\sigma_1}{\sigma} + \frac{\sigma_3}{\sigma_s} > 1; \tag{9}$$

The results of the calculations are given in figures 6 – 8.

By analysing the results obtained, we can conclude that the maximum stresses occur in the upper part of the pad in the contact area between the back plate and the side plate; they amount to 14.9 MPa and do not exceed the permissible values (15 MPa [22]).

3. The research into the thermal stress state of the composite brake pad with dual wedge-shaped wear

The strength of the brake pad was calculated and its dual wedge-shaped wear was taken into account. The calculation was made for the composite brake pad with dual wedge-shaped wear used for the wagon with a mileage of 63400 km. According to the measurements (figure 9, a), the

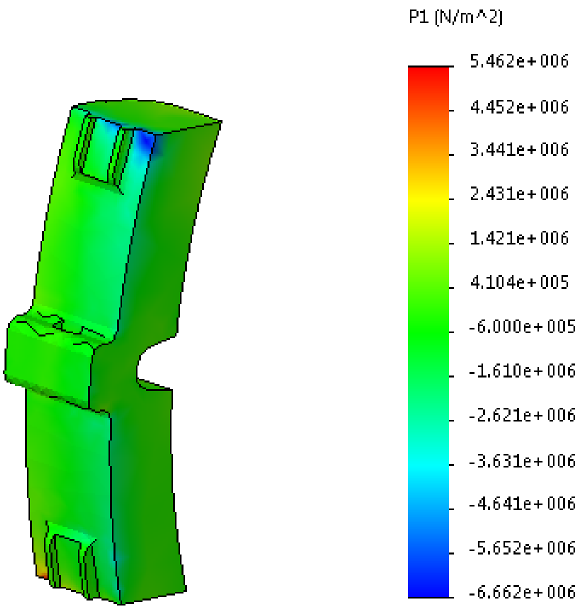

Figure 6. First principal stress in the pad.

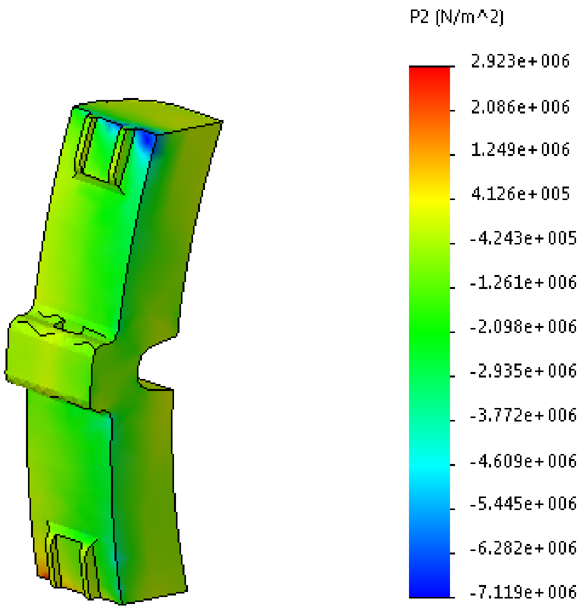

Figure 7. Second principal stress in the pad.

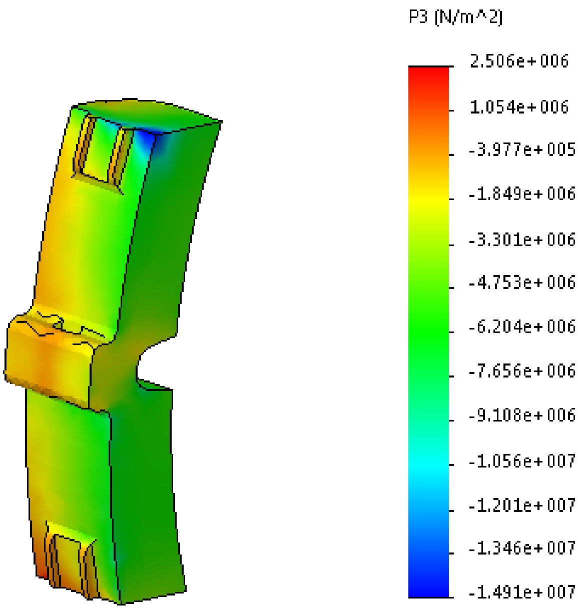

Figure 8. Third principal stress in the pad.

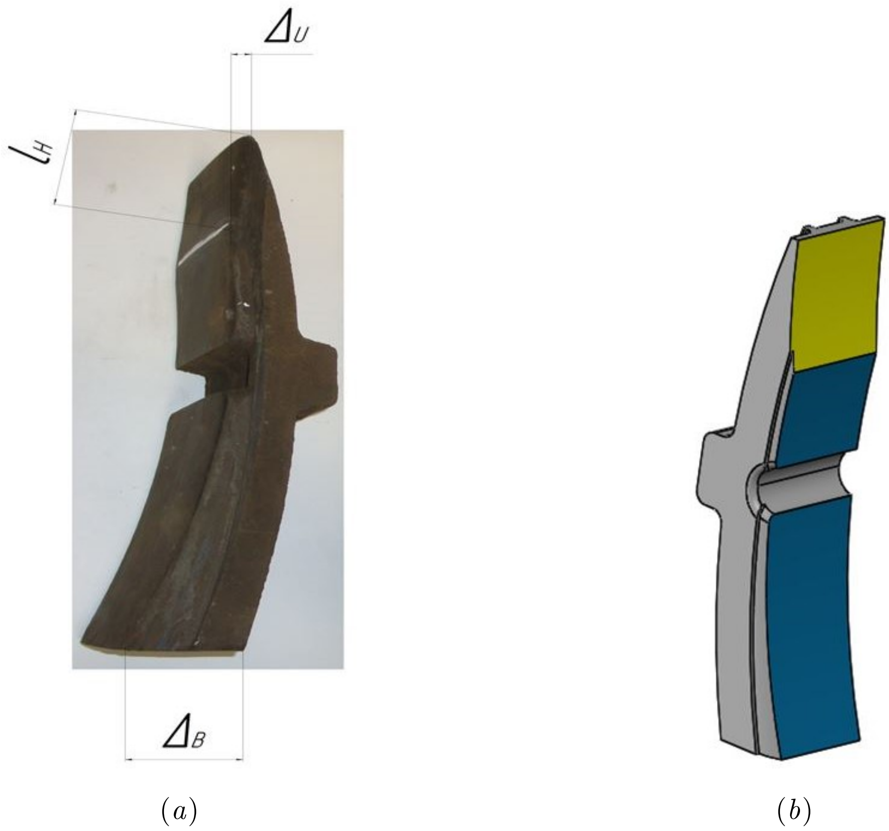

Figure 9. General view of the composite brake pad with dual wedge-shaped wear: (*a*) measurements of the geometric parameters of the pad to determine abnormal wear; (*b*) spatial model.

pad had the following abnormal wear characteristics: thickness at the upper end $\Delta_U = 10$ mm; thickness along the borderline of the plates $\Delta_{BL} = 27$ mm; thickness at the bottom end $\Delta_B = 20$ mm and length of harmful abrasion at the top of the pad $l_H = 85$ mm.

The design diagram of the pad included the loads identical to those shown in figure 10 and figure 11.

The finite-element model of the brake pad with wear consisted of 5429 elements and 24502 nodes (figure 12). The maximum element size of the mesh was 12 mm and the minimal element size was 2.4 mm. The number of elements in the circle was 9. The element size gain ratio was

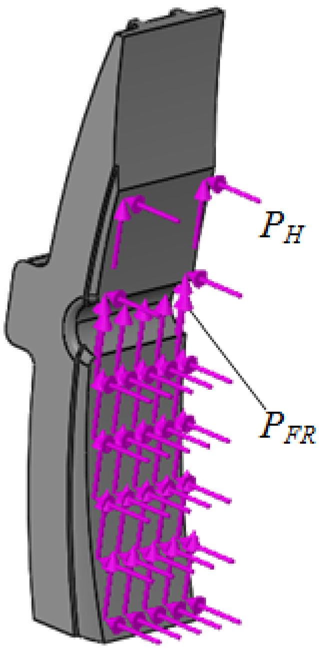

Figure 10. Design diagram of the pad with wear.

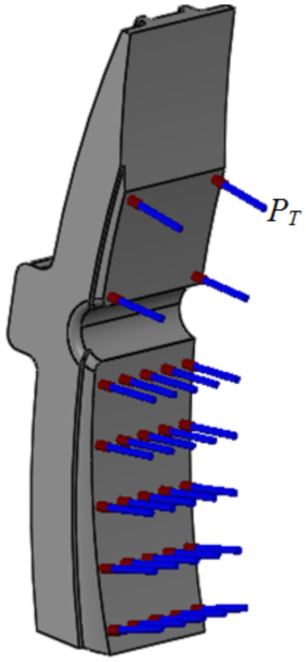

Figure 11. Diagram of the temperature load applied to the pad with wear.

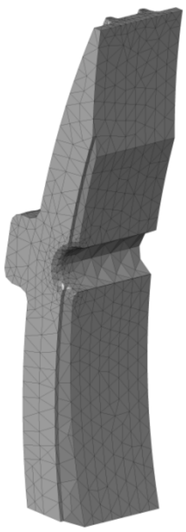

Figure 12. Finite-element model of the brake pad with wear.

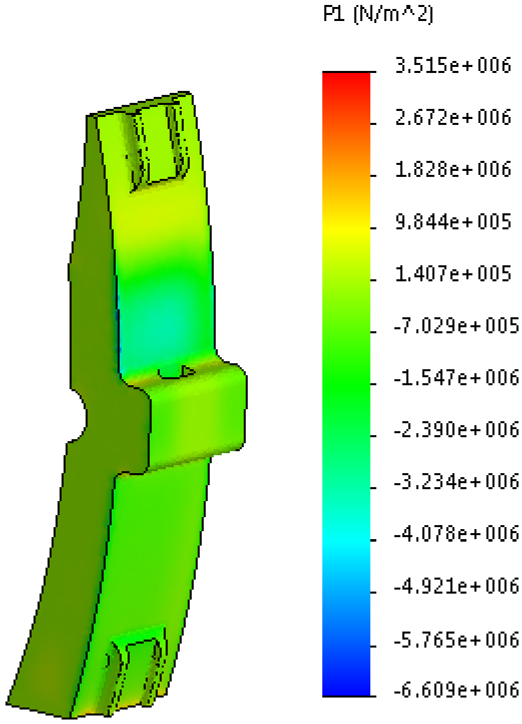

Figure 13. First principal stress in the pad with wears.

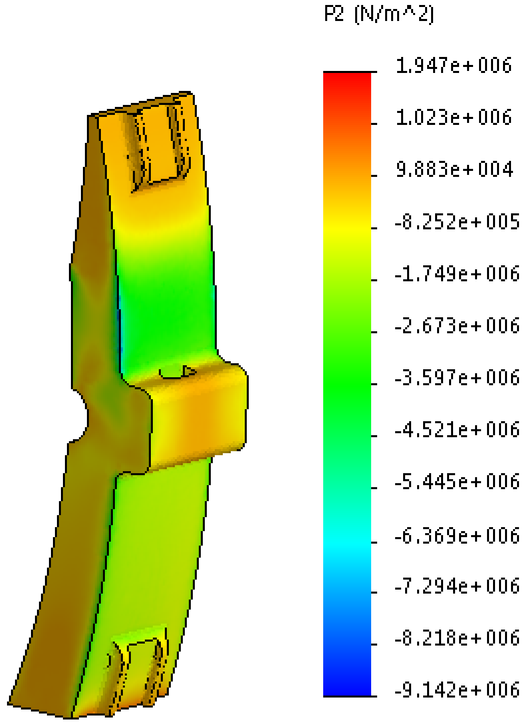

Figure 14. Second principal stress in the pad with wears.

1.6.

The results of the calculations are given in figures 13 – 15. The maximum stresses were recorded in the back plate of the pad and amounted to 18.7 MPa (third principal stress); they exceeded the permissible values by 19.8%.

The distribution of stresses along the upper part of the brake pad is shown in figure 16.

The graph presents the stresses by module. Their numerical values were determined using the probing option in SolidWorks Simulation. From this figure it can be concluded that the maximum stresses occur at a height of 135 – 140 mm from the top of the pad.

4. Conclusions

The research deals with the thermal stress state of the composite brake pad with rated parameters. It has been found that the maximum stresses (third principal stress) occur in the upper part of the pad in the contact area between the back plate and the side plate; they amount to 14.9 MPa and do not exceed the permissible values (15 MPa). The maximum stress distribution in the pad is explained by the fact that the frictional force is taken as upward in the calculation. When pointing it downward, which may be typical for the opposite triangle of the bogie, the dislocation of the stresses is opposite to the resulting loading diagram.

The research includes the thermal stress state of the composite brake pad with dual wedge-shaped wear. The maximum stresses occur in the back plate of the pad and amount to 18.7 MPa (third principal stress); they exceed the permissible values by 19.8%. This is explained by the fact that the useful area of the pad decreases, and, accordingly, its loading increases.

The research conducted proves the negative impact of dual wedge-shaped wear not only on the braking efficiency, but also on the strength of brake pads. This requires the development of measures aimed at eliminating this wear.

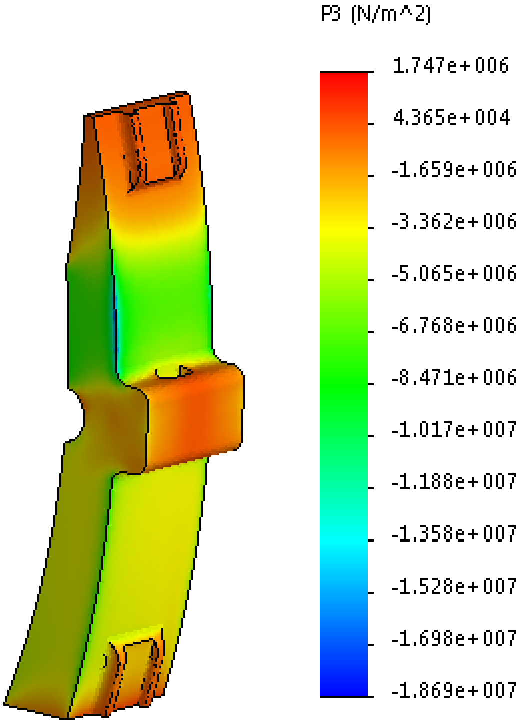

Figure 15. Third principal stress in the pad with wears.

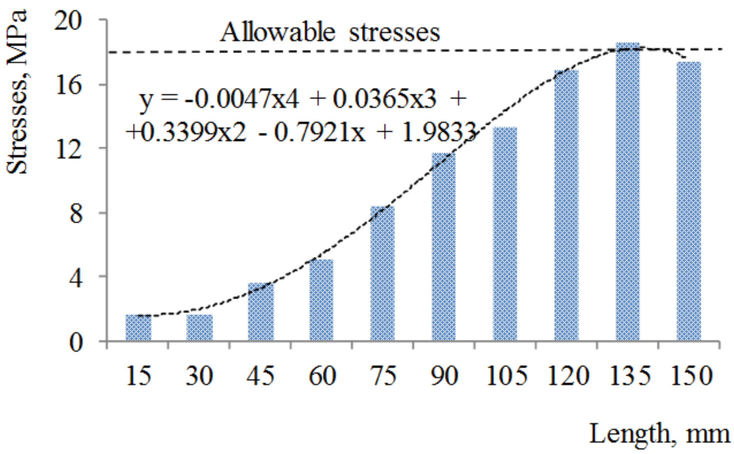

Figure 16. Distribution of stresses along the upper part of the brake pad.

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